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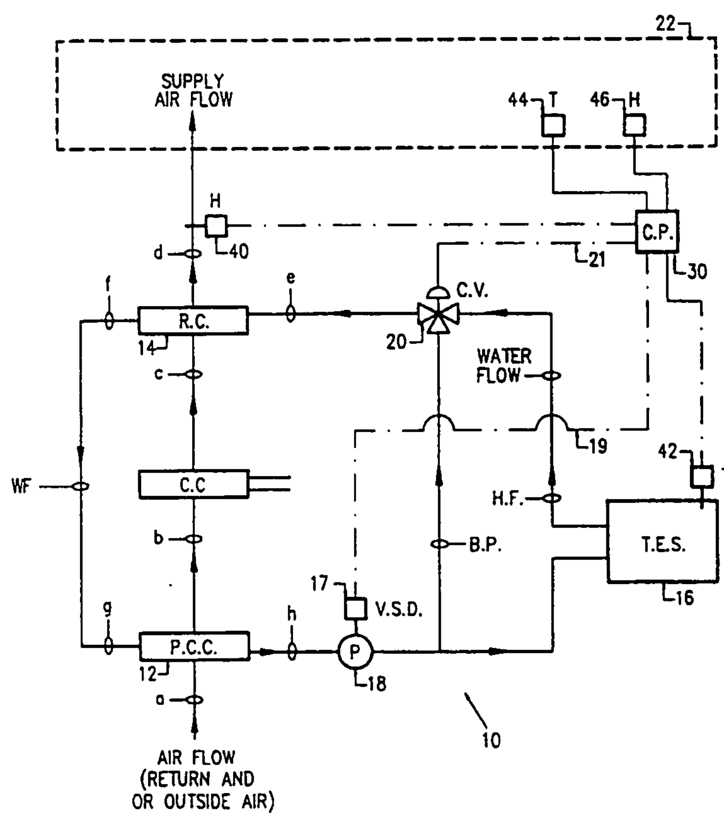
(54) Title: METHOD AND APPARATUS FOR LATENT HEAT EXTRACTION

(57) Abstract

A method and apparatus for improved latent heat extraction combines a run-around coil system (WF) with a condenser heat recovery system (56) to enhance the moisture removing capability of a conventional vapor compression air conditioning unit (50). The run-around coil system (WF) exchanges energy between the return (a) and supply air (d) flows of the air conditioning unit (10). Energy recovered in the condenser heat recovery system (56) is selectively combined with the run-around system energy extracted from the return air flow (a) to reheat the supply air (d) stream for downstream humidity control. A control system (30) regulates the relative proportions of the extracted return air flow energy and recovered heat energy delivered to the reheat coil (14) for efficient control over moisture in the supply air flow (d). Auxiliary energy in the form of electric heat energy (60) is further added to the recovered heat energy for additional reheat use.



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METHOD AND APPARATUS FOR LATENT HEAT EXTRACTION

Background of the Invention

This application pertains to the art of air conditioning methods and apparatus. More particularly, this application pertains to methods and apparatus for efficient control of the moisture content of an air stream which has undergone a cooling process as by flowing through an air conditioning cooling coil or the like. The invention is specifically applicable to dehumidification of a supply air flow into the occupied space of commercial or residential structures. By means of selective combination of extracted return air flow heat energy and recovered refrigerant waste heat energy, the supply air flow is warmed using a reheat coil apparatus. The return air flow entering the air conditioning coil is precooled with a precooling coil in operative fluid communication with the reheat coil. Heating of the occupied space may be effected using the combined reheat and precooling coils in conjunction with an alternative heat source such as electric, solar, or the like and will be described with particular reference thereto. It will be appreciated, though, that the invention has other and broader applications such as cyclic heating applications wherein a supply air flow is heated at the reheat coil irrespective of the instantaneous operational mode of the refrigerant system through the expedient of a thermal energy storage tank or the like.

Conventional air conditioning systems use a vapor compression refrigeration cycle that operates to cool an indoor air stream through the action of heat transfer as the air stream comes in close contact with evaporator type or flooded coil type refrigerant-to-air heat exchangers or coils. Cooling is accomplished by a

reduction of temperature as an air stream passes through the cooling coil. This process is commonly referred to as sensible heat removal. A corresponding simultaneous reduction in the moisture content of the air stream typically also occurs to some extent and is known as latent heat removal or more generally called dehumidification. Usually the cooling itself is controlled by means of a thermostat or other apparatus in the occupied space which respond to changes in dry bulb temperature. When controlled in this manner, dehumidification occurs as a secondary effect incidental to the cooling process itself. As such, dehumidification of the indoor air occurs only when there is a demand for reduced temperature as dictated by the thermostat.

To accomplish dehumidification when the thermostat does not indicate a need for cooling, a humidistat is often added to actuate the air conditioning unit in order to remove moisture from the cooled air stream as a "byproduct" function of the cooling. In this mode of operation, heat must be selectively added to the cooled air stream to prevent the conditioned space from over-cooling below the dry bulb set point temperature. This practice is commonly known as "reheat".

Many sources of heat have been used for reheat purposes, such as hydronic hot water with various fuel sources, hydronic heat recovery sources, gas heat, hot gas or hot liquid refrigerant heat, and electric heat. Electric heat is most often used because it is usually the least expensive alternative overall. However, the use of electric heat to provide the reheat energy is proscribed by law in some states, including Florida for example.

In order to conserve energy, it has been suggested that recovered heat be used as a source for the reheat. Accordingly, one method to improve the moisture removal capacity of an air conditioning unit, while simultaneously providing reheat, is to provide two heat exchange surfaces each in one of the air streams entering or leaving the cooling coil while circulating a working fluid between the two heat exchangers. This type of simple system is commonly called a run-around system.

Run around systems have met with limited success. The working fluid is cooled in a first heat exchange surface placed in the supply air stream called a reheat coil. The cooled working fluid is then in turn caused to circulate through a second heat exchange surface placed in the return air stream called a precooling coil. This simple closed loop circuit comprises the typical run-around systems available heretofore.

The precooling coil serves to precool the return air flow prior to its entering the air conditioning cooling coil itself. The air conditioning coil then provides more of its cooling capacity for the removal of moisture from the air stream otherwise used for sensible cooling. However, the amount of reheat energy available in this process is approximately equal to the amount of precooling accomplished. This is a serious constraint. Additional reheat energy is often needed for injection into the run-around system to maintain the desired dry bulb set point temperature and humidity level in the conditioned space. As described above, supplemental electric reheat has been used with some success.

In addition, the growth of molds in low velocity air conditioning duct systems has recently become a major indoor air quality concern. One of the control measures recognized as having the capability of limiting this undesirable growth is the maintenance of the relative humidity at 70 percent or lower in the air conditioning system air plenums and ducts. Within limits, reheat can be used to precisely control the relative humidity. However, as described above, the amount of reheat energy from the run-around systems available today may not be sufficient to consistently provide the above level of humidity control, particular during periods of operation when the air temperature entering the precooling coil is lower than the system design operating temperature.

As a further complication, air conditioning units are also often used for heating purposes as well as for cooling and dehumidification. Electric heating elements are often provided in the air conditioning units to selectively provide the desired amount of heat at precise times of the heating demand. The above demand for heating energy will most often correspond with the demand for heating at other air conditioning units in the locality. This places a substantial and noticeable demand on the electrical power utility system in the community. In many areas, this peak demand has exceeded the capacity of the power system. The electric utility companies have responded with incentives encouraging their customers to temper their demand during regional peak demand periods. These incentives are often in the form of demand charges which encourage the customer to reduce their demand on the system at those times in order to avoid incremental costs in addition to the regular base rates.

It has, therefore, been deemed desirable to provide an economical solution that meets the various needs of air conditioning system installation requirements while also operating in compliance with current and projected local environmental and energy-related laws.

Summary of the Invention

This invention improves the dehumidification capabilities of conventional air conditioning systems through the addition of a run-around system having a supplemental heat energy source for reheat use. The amount of reheat energy that can be incrementally added to the stream air leaving the conditioning unit is thereby increased. An air conditioning unit so configured is capable of operating continuously over a wide range of conditions for providing dehumidification to the occupied space independent of the sensible cooling demand at the conditioned space. Such a system is further capable of maintaining a precise relative humidity level in the air conditioning duct system to enhance the indoor air quality of the occupied conditioned space. Further, the overall system may be used to heat the occupied space through the expedient of the stored energy scheme according to the teachings of the preferred embodiments.

In the preferred embodiment, the supplemental heat source is heat recovered from the refrigeration process of the particular installed air conditioning system having the reheat requirement. In another embodiment, the supplemental heat is an alternative energy source, such as a gas or electric boiler, or water heater. The new energy source may be of

particular benefit for use with an air conditioning system that uses chilled water or cold brine for the cooling medium.

The basic preferred embodiment of the invention comprises heat exchange coils in the entering air stream and leaving air stream of an air conditioning unit primary cooling coil. The basic preferred embodiment further comprises a circulating pump, and a supplementary heat source, which can be a heat recovery device on the air conditioning unit refrigeration circuit or a conventional liquid heater or the like.

Brief Description of the Drawings

FIGURE 1 illustrates a schematic view of the preferred embodiment of the apparatus for latent heat extraction according to the invention;

FIGURE 2 illustrates a schematic view of the preferred embodiment of the invention when used with a conventional air conditioning unit having a vapor compression type refrigeration system;

FIGURE 3 illustrates a schematic of the preferred embodiment of the invention when used with an air conditioning unit using chilled water for the cooling medium;

FIGURES 4a, 4b are flow charts of the control procedure executed by the control apparatus during the space cooling mode of operation;

FIGURES 5a, 5b are flow charts of the control procedure executed by the control apparatus during the space dehumidification mode of operation;

FIGURE 6 is a flow chart of the control procedure executed by the control apparatus during the space heating mode of operation;

FIGURE 7 is a flow chart of the control procedure executed by the control apparatus during the various operational modes for maintenance of the thermal energy storage tank temperature;

FIGURE 8 is a coil graph of a first sample calculation;

FIGURE 9 is a coil graph of a second sample calculation;

FIGURE 10 is a coil graph of a third sample calculation; and,

FIGURE 11a, 11b are a psychometric chart of the combined first, second and third sample calculations and a protractor for use with the psychometric chart.

Detailed Description of the Preferred Embodiments

Referring now to the drawings wherein showings are for purposes of illustrating the preferred embodiments of the invention only and not for purposes of limiting same, the FIGURES show a moisture control apparatus **10** for conditioning the air in an occupied space **22**. The apparatus **10** comprises suitably arranged components including a precooling coil **12** in a return air flow **a,b**, a reheat coil **14** in a supply air flow **c,d**, a thermal energy storage tank **16** operatively associated with a source of heat, a working fluid pump **18** for circulating a working fluid **WF** through a series arrangement of the above coils, a variable speed drive **17** for controlling the speed of pump **18** and a modulated control valve **20** for metering the working fluid. An apparatus controller **30** directly modulates the control valve **20** and generates variable speed command signals for control over the working fluid pump **18**.

With particular reference to FIGURE 1, the working fluid **WF** enters the control valve **20** from one of

two sources including a bypass fluid flow **BP** and a heated fluid flow **HF**, the latter passing first through the thermal energy storage tank **16**. In both above cases, the flow of the working fluid is motivated by the working fluid pump **18**. A mixture of bypass fluid flow **BP** and heated fluid flow **HF** may be accomplished over a continuum by a blending control valve substituted for the modulated control valve **20**, along with an analog output signal from the apparatus controller **30** described below.

The apparatus controller **30** is an operative communication with a plurality of system input devices, each of which sense various physical environmental conditions. These input devices include a supply airflow humidity sensor **40**, a thermal energy storage tank temperature sensor **42**, an occupied space dry bulb temperature sensor **44**, and an occupied space humidity sensor **46**. The humidity sensor **40** may be replaced with a temperature sensor for ease of maintenance and reliability.

In addition, the controller **30** is in operative communication with a plurality of active output devices. The output devices are responsive to signals deriving from the apparatus controller **30** according to programmed control procedures detailed below. In the preferred embodiment, the output devices comprise the control valve **20** responsive to a control valve signal **21**, and a variable speed drive **17** responsive to a pump speed command signal **19**. Additional input and output signals, including alarm and data logging signals or the like, may be added to the basic system illustrated in FIGURE 1 as understood by one skilled in the art after reading and understanding the instant detailed description of the preferred embodiments.

With particular reference now to FIGURE 2, a schematic diagram of the preferred embodiment of the apparatus of the invention is illustrated adapted for use with a conventional air-conditioning unit having a vapor compression type refrigeration system. The system includes a compressor **50** for compressing a compressible fluid **CF** and a condenser coil **52**. An evaporative cooling coil **54** absorbs heat from a return air flow **a**, **b** resulting in a cooled supply air flow **c**, **d** into an occupied space **22**. These various air conditioning components may be assembled in a single package, known in the art as a roof-top unit, or may be provided as a system comprising separated items, such as what is called a split system.

With continued reference to FIGURE 2, a reheat coil **14**, as described above, is placed in the supply air flow **c**, **d** after (downstream of) the evaporative cooling coil **54**, while a precooling coil **12** is placed in the return air flow **a**, **b** before (upstream of) the cooling coil **54**. For full effectiveness of the air quality control measure of the instant invention, the reheat coil **14** should be physically mounted as close as possible to the cooling coil **54**. The precooling coil **12** can be mounted in any convenient location and may be so situated as to precool only the outside air, only the return air, or a mixture of the outside air and return air (not shown).

As above, the working fluid pump **18** is connected to a variable speed drive **17** which operates to circulate the working fluid **WF** between the reheat coil **14**, the precooling coil **12**, and the thermal energy storage tank **16**. In the preferred embodiment, the working fluid is water. In general, the overall system may be used in various operating modes including a space

cooling mode, a space dehumidification mode, and a space heating mode. To describe the full operation of the system, each of the operational modes will be described in detail below.

In the space cooling mode, the working fluid pump **18** operates when the refrigeration system compressor **50** is operating. In this mode, the compressor **50** is responsive to the occupied space dry bulb temperature sensor **44**. The pump **18** is driven by the variable speed drive **17** which regulates the water flow to maintain the desired humidity setting at the supply air flow humidity sensor **40**. Water flow is increased on a rise in the relative humidity above a predetermined set point and conversely decreased on a drop in relative humidity at the supply air flow humidity sensor **40** below said set point.

In the space dehumidification mode, the compressor **50** of the conventional air-conditioning unit is operated to maintain the humidity at the occupied space **22**, as sensed by the occupied space humidity sensor **46**, the speed of the working fluid pump **18** is regulated to maintain the desired temperature of the occupied space **22** as sensed by the occupied space dry bulb temperature sensor **44**. In this dehumidification mode of operation, working fluid flow **WF** is increased on a drop in temperature at the occupied space dry bulb temperature sensor **44**, and water flow is conversely decreased on a rise in the occupied space temperature. Responsive to command signals from the apparatus controller **30** and according to the control algorithms detailed below. When the temperature in the occupied space is a controlling factor in setting the working fluid pump speed, the supply air flow humidity set point is used to establish at a minimum working fluid pump

speed. In any of the above modes, working fluid flow control may be accomplished using a two-port valve with a modulating actuator in place of the variable speed drive **17**.

In general terms, cooled air leaving the evaporative type cooling coil **54** enters the reheat coil **14** where it absorbs heat from the working fluid flow in the tubes of the reheat coil itself. There is a drop in heat content of the working fluid from points **e** to **f** equal to the rise in the heat content of the air stream from points **c** to **d**. The working fluid is transferred through the piping system **32** to the precooling coil **12**. Cooled working fluid from the reheat coil **14** absorbs heat from the return air flow stream as the air passes over the precooling coil surfaces. There is a rise in the heat content in the working fluid from points **g** to **h** equal to the drop in the heat content of the air stream from points **a** to **b**. These principles are each generally well-known and established in the art.

Heat exchange pump **58** operates when the compressor **50** is operating and when the temperature and the thermal energy storage tank **16** is below a predetermined set point at the thermal energy storage tank temperature sensor **42**. The function of the heat exchange pump **58** is to transfer working fluid heated by the hot refrigerant gas in a heat exchanger **56**. The heat exchange pump **58** stops even though the compressor **50** is running when the temperature in the thermal energy storage tank **16** is at an upper working fluid temperature set point as determined by the thermal energy storage tank temperature sensor **42**. The general function of the heat exchanger **56** is to provide supplemental heat to charge the thermal energy storage tank **16** with hot working fluid for heating and/or reheat operation.

An electric heating element **60** may be used as an additional energy source to heat the working fluid when there is a demand for more heat than may be provided by the heat exchanger **56**. The supplemental electric heating operation is controlled by the apparatus controller **30** to operate as a secondary source of energy when the temperature in the thermal energy storage tank **16** drops below the desired set point as determined by the thermal energy storage tank temperature sensor **42**. As an example, if the desired minimum temperature in the thermal energy storage tank is 120°F and the desired maximum temperature is 125°F, the heat exchange pump **58** is made to begin operation on a drop in temperature below 120°F. Conversely, when the thermal energy storage tank temperature drops to 120°F, the electric heating element **60** is activated by the apparatus controller **30**. On a rise in the thermal energy storage tank temperature, the heating element **60** is first turned off, and on a continued rise in temperature to the 125°F set point, the heat exchange pump **58** is next turned off. This scheme is hierarchically arranged in order to conserve energy by first recovering energy from the air-conditioning unit which might otherwise be lost.

Multiple heating elements similar to the electric heating element shown may be provided and controlled by a step controller to match the energy input to the heating load in stages of electric heat. An SCR controller may be used to proportionally control the amount of heat energy added to the thermal energy storage tank **16** as a function of the tank temperature differential from minimum to maximum set points. On a larger scale, such as neighborhood-wide, the electric heating controls may be circuited to allow the lock-out

of the electric heating elements during periods of peak electrical demand throughout the neighborhood. This lock-out control may be in the form of an external signal, such as may be provided from the neighborhood power company, or from the owner's energy management system. The control may further be obtained from a signal from the system controls contained in the apparatus controller **30**, as a function of the time of day, demand limiting, or other energy management strategies.

Referring next to FIGURE 3, a schematic diagram of the preferred embodiment of the invention is illustrated and modified for use with an air-conditioning unit using chilled water as the cooling medium. The chilled water system uses a chilled water cooling coil **70** which may be mounted in a duct or plenum, or can be mounted in an air-handling unit with integral or remote mounted fans. Chilled water systems are usually provided with a control valve **72** to regulate the amount of cooling accomplished by the system in response to the occupied space dry bulb temperature sensor **44**.

With continued reference to FIGURE 3, a reheat coil **14**, as described above, is placed in the supply air flow **c,d** after the evaporative cooling coil **54**, while a precooling coil **12** is placed in the return air flow **a,b** before the cooling coil **54**. For full effectiveness of the air quality control measure of the instant invention, the reheat coil **14** should be mounted as close as possible to the cooling coil **54**. The precooling coil **12** can be mounted in any convenient location and may be so situated as to precool only the outside air, only the return air, or a mixture of the outside air and return air (not shown).

The pump **18** is connected to a variable speed drive **17** which operates to circulate the working fluid **WF**, in this preferred embodiment water, between the reheat coil **14**, the precooling coil **12**, and the thermal energy storage tank **16**. In general, the overall system may be used in various operating modes including a space cooling mode, a space dehumidification mode, and a space heating mode. To describe the operation of the system, each of the operational modes will be introduced here and described in detail below.

In the space cooling mode, the working fluid pump **18** operates when there is a demand for cooling in space **22**. In this mode, the control valve **72** is responsive to the occupied space dry bulb temperature sensor **44**. The pump **18** is driven by the variable speed drive **17** which regulates the water flow to maintain the desired humidity setting at the supply air flow humidity sensor **40**. Water flow is increased on a rise in the relative humidity above a predetermined set point and conversely decreased on a drop in relative humidity at the supply air flow humidity sensor **40** below said set point.

In the space dehumidification mode, the air-conditioning unit is operated to maintain the humidity at the occupied space **22**, as sensed by the occupied space humidity sensor **46**, the speed of the working fluid pump **18** is regulated to maintain the desired temperature of the occupied space **22** as sensed by the occupied space dry bulb temperature sensor **44**. In this dehumidification mode of operation, working fluid flow **WF** is increased on a drop in temperature at the occupied space dry bulb temperature sensor **44**, and water flow is conversely decreased on a rise in the occupied space temperature. Responsive to command signals from the

apparatus controller **30** and according to the control algorithms detailed below. When the temperature in the occupied space is a controlling factor in setting the working fluid pump speed, the supply air flow humidity set point is used to establish at a minimum working fluid pump speed. In any of the above modes, working fluid flow control may be accomplished using a two-port valve with a modulating actuator in place of the variable speed drive **17**.

In general terms, cooled air leaving the type cooling coil **70** enters the reheat coil **14** where it absorbs heat from the working fluid flow in the tubes of the reheat coil itself. There is a drop in heat content of the working fluid from points **e** to **f** equal to the rise in the heat content of the air stream from points **c** to **d**. The working fluid is transferred through the piping system **32** to the precooling coil **12**. Cooled working fluid from the reheat coil **14** absorbs heat from the return air flow stream as it passes over the precooling coil surfaces. There is a rise in the heat content in the working fluid from points **g** to **h** equal to the drop in the heat content of the air stream from points **a** to **b**. These principles are each generally well-known and established in the air.

An electric heating element (not shown) may be used as a supplemental energy source to heat the working fluid when there is a demand for additional heat. The supplemental electric heating operation is controlled by the apparatus controller **30** to operate as a secondary source of energy when the temperature in the thermal energy storage tank **16** drops below the desired set point as determined by the thermal energy storage tank temperature sensor **42**. As an example, if the desired minimum temperature in the thermal energy storage tank

is 120°F and the desired maximum temperature is 125°F, the electric heating element (not shown) is activated by the apparatus controller **30** when the thermal energy storage tank temperature drops to 120°F. On a return in the thermal energy storage tank temperature to 125°F, power to the heating element is turned off.

Multiple heating elements similar to the electric heating element described above may be provided and controlled by a step controller to match the energy input to the heating load in stages of electric heat. An SCR controller may be used to proportionally control the amount of heat energy added to the thermal energy storage tank **16** as a function of the tank temperature differential from minimum to maximum set points. On a larger scale, such as neighborhood-wide, the electric heating controls may be circuited to allow the lock-out of the electric heating elements during periods of peak electrical demand throughout the neighborhood. This lock-out control may be in the form of an external signal, such as may be provided from the neighborhood power company, or from the owner's energy management system. The control may further be obtained from a signal from the system controls contained in the apparatus controller **30**, as a function of the time of day, demand limiting, or other energy management strategies.

With reference now to FIGURES 2, 3, 4a and 4b, the control method for the space cooling mode operation will be described. In the space cooling mode, the compressor **50** of FIGURE 2 and the chilled water cooling coil **70** of FIGURE 3 are operated **104**, **106** to maintain the desired set point dry bulb temperature in the occupied space **22** according to the occupied space dry bulb temperature sensor **44**. In the conventional air-

conditioning system, the compressor **50** starts **106** on a rise in occupied space temperature above a predetermined set point and stops **104** on a fall in occupied space temperature below the set point temperature **102** as sensed by the occupied spaced dry bulb temperature sensor **44**. Correspondingly, in the chilled water system, the control valve **20** opens **106** on a rise in the occupied space temperature and closes **104** on a fall in the occupied space temperature below the predetermined set point at occupied space dry bulb temperature sensor **44**. In either case, the speed of the working fluid pump **18** is regulated by the variable speed drive **17** to maintain the desired relative humidity **110** in the supply air flow **d** as sensed by the supply air flow humidity sensor **40**.

The pump speed is also controlled to maintain the desired relative humidity **108** in the occupied space **22** according to the occupied space humidity sensor **46**. The working fluid pump speed increases **114** on a rise in the relative humidity above the supply air or the occupied space air relative humidity set points. The working fluid pump speed decreases **112** on a fall in the relative humidity below the set points.

When the variable speed drive **17** is at full speed **118**, the control valve **20** is modulated to maintain the desired humidity set points **120, 122**. The control valve **20** is positioned to bypass the thermal energy storage tank **16** when the working fluid pump **18** is operating at speeds of less than 100% of full speed. When the variable speed pump **18** is at full speed, the control valve **20** is modulated open **126** to thermal energy storage tank **16** on a rise in supply air **122** or occupied space **120** relative humidity above the predetermined set points according to the supply air flow humidity sensor

40 and the occupied space humidity sensor **46** respectively. In this state, the working fluid flows to the reheat coil **14** directly from the thermal energy storage tank **16** as a heated working fluid flow **HF**. The control valve **20** is modulated closed **124** on a decrease in the supply air or occupied space air relative humidity below the predetermined set points.

Next, with reference to FIGURES 2, 3, 5a and 5b, the control method for the space dehumidification operating mode will now be described. During this mode, when the occupied space dry bulb temperature set point is satisfied according to the occupied space dry bulb temperature sensor **44**, the compressor **50** of the conventional air conditioning unit is operated to maintain the desired occupied space relative humidity. In the chilled water system, the chilled water control valve **72** is operated to maintain the desired occupied space relative humidity. In this mode, the compressor **50** or the chilled water control valve **72** operate **208** on a rise in the occupied space relative humidity **202** above the set point and stop **206** on a drop in the occupied space relative humidity **202** below said set point. The working fluid pump **18** and control valve **20** are controlled **210-222** according to the space cooling mode described above.

With reference next to FIGURES 2, 3 and 6, the control method for the space heating operating mode will now be described. In this mode, the thermal energy storage tank **16** is utilized to maintain the desired occupied space dry bulb temperature according to the physical conditions sensed by the occupied space humidity sensor **46**. Normally in this mode, the compressor **50** and chilled water control valve **72** are both off in the standard air-conditioning system and

chilled water systems respectively. In the instant space heating mode, the working fluid **WF** is circulated exclusively through the thermal energy storage tank **16** as a heated fluid flow **HF**. No flow is permitted through the bypass as a bypass fluid flow **BP**. This is accomplished via the control valve **20** modulated open **302** according to the control valve signal **21** from the apparatus controller **30**. The speed of the working fluid pump **18** is adjusted **306, 308** to maintain the desired temperature set point **304** in the occupied space **22**. As an alternative means, the working fluid pump **18** may be continuously operated, but cycled on and off according to the demand for heating as sensed by the occupied space dry bulb temperature sensor **44**. This results in an average heating defined by the duty cycle of the alternating on/off cycles.

With reference now to FIGURE 7, the thermal energy storage tank maintenance routine **TES** will be now described in detail. The method is a subroutine in each of the space cooling, space dehumidification, and space heating control methods described above. In this control subroutine procedure, heat exchange pump **58** operates **408** when the compressor **50** is operating **402** and when the temperature in the thermal energy storage tank **16** is below the set point **404** at temperature sensor **42**. The function of pump **58** is to transfer water **WF** heated by the hot refrigerant gas in the heat exchanger **56**. The pump stops **406** when the temperature in the tank is at the upper water temperature set point **404** at the temperature sensor **42**. The function of the heat exchanger is to provide supplemental heat to charge the thermal storage tank **16** with hot water for heating and/or reheat operation.

Electric heating element **60** may be used as an additional energy source to heat the water when there is a demand for more heat than can be provided by the heat exchanger. The electric heating operation is controlled by the apparatus controller **30** to operate **414** as the second source of energy when the temperature in the thermal storage tank **16** drops below the desired set point **410** at sensor **42**. As an example, if the desired minimum temperature in the tank is 120° F and the desired maximum temperature is 125° F, the pump **58** starts on a drop in temperature below 125° F. When the tank temperature drops to 120° F, the electric heating element **60** is activated. On a rise in tank temperature the heating elements are turned off first **416**, and on a continued rise in temperature to 125° F the pump **58** is, in turn, shut off **406**. Multiple heating elements may be provided and controlled by a step controller to match the energy input to the heating load in stages of electric heat or an SCR controller can be used to proportionately control the amount of heat energy added to the tank as a function of the tank temperature differential from minimum to maximum set points.

The electric heating controls may further be circuited to allow for a lock out **416** of the electric heating elements during periods of peak community electrical demand **412**. This lock out control could be provided from an external signal such from the power company or from the owner's energy management system. The control could be from a signal from the system controls contained in control **30** as a function of time of day, demand limiting, or other energy management strategies.

With reference once again to FIGURE 2, the system may be operated in a variety of modes. In

general, when the overall system is operating in either the cooling mode or the dehumidifying mode the cold air leaving the evaporator coil **50** enters the reheat coil **14** where it absorbs heat from the moving water stream **WF** in the tubes of the reheat coil **12**. There is a corresponding drop in the heat content of the circulating water from points **e** to **f** equal to the rise in heat content of the air stream from points **c** to **d**. The water **WF** is transferred through a piping conduit system to the precooling coil. Cold water entering the precooling coil **12** absorbs heat from the return air stream **a** as it passes over the coil surfaces. There is a rise in heat content of the circulating water from points **g** to **h** equal to the drop in heat content of the air stream from points **a** to **b**. Representative sample calculations follow below.

SAMPLE CALCULATIONS

The sample calculation **A** immediately below is illustrated in the coil graph of FIGURE 8 and in the psychometric chart of FIGURES 11a, 11b wherein it is Given that:

- Required indoor temperature is 75°F at 45% relative humidity;
- Indoor cooling load (peak load) is

220.0	MBTU/Hour Sensible
94.3	MBTU/Hour Latent
314.3	MBTU/Hour Total;

- Outdoor air temperature at peak cooling load is 93°F dry bulb and 76° dry wet bulb;
- Amount of ventilation air (outside air) required is 2500 CFM;
- Desired supply air relative humidity level is 70% maximum;

- Return air heat gain assumed equal to a 2°F ΔT rise; and
- Fan and motor heat gain assumed equal to a $1\frac{1}{2}$°F ΔT rise.

Statement of Solution:

- Sensible heat ratio = $\frac{220.0}{314.3} = 0.70$
- Room condition line intersects 70% RH line at 55°F.
- Supply air volume required:

  $V = \frac{220000\ BTU/HR}{1.1 \cdot 20°\ \Delta T}$
- Reheat energy required to provide 70% Rel. Hum. in supply air stream:

  $Q = 10000\ CFM \cdot 1.1 \cdot [(55-47)°F\Delta T - 1\frac{1}{2}°F)]$

  $= 71500\ BTU/HR$
- Water flow rate required through reheat coil assuming $6\frac{1}{2}$°F ΔT and 12°F approach temperature:

  $V = 71500\ BTU/Hour/(500 \cdot 6.5°F\ \Delta T) = 22\ GPM$
- Coil conditions - Temperature:

	Air	Water
Entering Coil	47	65.5
Leaving Coil	53.5	59.0

- Precooling coil air temperature drop (sensible cooling):

  $\Delta T = \frac{Q}{1.1 \cdot CFM}$

  Q = Amount of energy recovered for supply air stream at reheat coil

  $\Delta T = 71500\ BTU/Hour/1.1 \cdot 10000CFM = 6.5°F\ \Delta T$
- Coil conditions - Temperature

	Air	Water
Entering Coil	81	59
Leaving Coil	74.5	65.5

The sample calculation B immediately below is illustrated in the coil graph of FIGURE 9 and in the psychometric chart of FIGURES 11a, 11b wherein it is

<u>Given that:</u>

- Same condition as calculation (A), except indoor sensible cooling load is 110.0 MBTU/Hour; and,
- Assume supply air dew point is fixed at 45°F due to coil characteristics;

<u>Statement of Solution:</u>

-New sensible heat ratio

$$\frac{110.0}{110 + 94.3} = 0.54$$

- Reheat energy required

$Q = 10000\ \text{CFM} \cdot 1.1 \cdot [(65-47)°F\Delta T - 1\tfrac{1}{2}°F)]$
$= 181500\ \text{BTU/hour}$

- Water temperature required using 22 GPM flow rate

$$\Delta T = \frac{181500\ \text{BTU/hour}}{22\text{GPM} \cdot 500} = 16.5°\ \Delta T\ °F$$

- Reheat energy required from refrigerant heat recovery:

$Q_3 = Q_1 - Q_2$

$Q_1 =$ Total reheat required

$Q_2 =$ Water heat gain in precooling coil (from Calculation (A))

$Q_3 = 181500 - 71500\ \text{BTU/hour} = 110{,}000\text{BTU/hour}$

- Temperature rise required by water through heat reclaim device:

$$\Delta T = \frac{Q_3}{500} \cdot 22\text{GPM} = \frac{110000}{500} \cdot 22\text{GPM} = 10°F$$

The sample calculation C immediately below is illustrated in the coil graph of FIGURE 10 and in the psychometric chart of FIGURES 11a, 11b wherein it is

Given that:

- Same conditions as Calculation (A), except:
- Space sensible cooling load is 110 MBTU/hour
- Refrigeration compressor(s) provided with capacity reduction to reduce amount of refrigerant flow, matching the new cooling load; this results in an increased dew point in the supply air.

Statement of Solution:

- Assuming capacity reduction raises the supply air dew point to 51°F;
- Space condition line intersects dew point line as 65°F db, this is the supply air dry bulb temperature; space condition line extends up and to the right, establishing a new room condition of 75°F at ~ 53% relative humidity.

The sample calculation immediately below illustrates the Heating Mode of operation wherein it is

<u>Given:</u>

- Space heating load is 216000 BTU/Hour, peak;
- Supply air volume is 10,000 CFM (from Calculation (A));
- Desired space temperature is 72°F;
- Outside air temperature is 35°F; and,
- Outside air volume is 2500 CFM.

<u>Statement of Solution:</u>

- Supply air temperature required is

$$T_s = \frac{72°F\ 216000\ BTU/hour}{1.1 \cdot 10000\ CFM} = 72°F + 20 = 92°F$$

- Mixed air temperature is:

$$T_m = 72°F \left[\frac{216000\ BTU/hour}{10000\ CFM} \cdot (72 - 35)°F \right]$$

$$= 62.75°F$$

- Total heating required

$$Q = 1.1 \cdot 10000\ CFM \cdot (92 - 62.75)°F$$

$$= 321750\ BTU/hour\ = 94KW$$

- Heat provided from thermal storage - ASSUMPTIONS: full heating shift to OFF peak, 10 hour heating period, 60% diversity.
- Heating required:

$$Q = 10\ hours \cdot 321750BTU/hour \cdot .6\ diversity$$

$$= 1930500\ BTU$$

- Heat input to thermal storage:
  - During moderate temperature periods recovered heat would be used to charge the storage tank. During cold weather, when the cooling system

is off, the electric heat would be used to store the energy.

- Electric heater size:

  $Q = 1930500$ BTU/14 hours = 137900 BTU/hour

  = 40 KW*

- Thermal storage volume required -

  ASSUMPTIONS: minimum useful temperature is 100°F and storage temperature is 140°F.

$$V = \frac{1930500\text{BTU}}{8.35\ \text{lb/gal.} \cdot 1\ \text{BTU/lb}\cdot{}^\circ\text{F} \cdot (140 - 100)^\circ\text{F}}$$

$V = 5780$ Gallons

- The amount of storage could be reduced if the electric heat is allowed to operate during the peak period (at a reduced rate to provide some demand saving):

$$V = \frac{1930500\text{BTU} - 10\text{hrs} \cdot 20\text{KW} \cdot 3413\ \text{BTU/KW}}{8.35\ \text{lb/gal.} \cdot 1\ \text{BTU/lb}\cdot{}^\circ\text{F} \cdot (140 - 100)^\circ\text{F}}$$

$V = 3736$ Gallons

* Heater size and/or storage volume would be increased slightly to account for system loses.

The invention has been described with reference to the preferred embodiments. Obviously modifications and alterations will occur to others upon a reading and understanding of this specification. It is my intention to include all such modifications and alterations insofar as they come within the scope of the appended claims and equivalents thereof.

Having thus described the invention, I now claim:

1. A moisture control apparatus for use with a fluid compression air conditioning system having a compressor for compressing a compressible fluid, and a cooling coil where the compressible fluid decompresses absorbing thermal energy from a return air flow as a cooled supply air flow, the moisture control apparatus comprising:

a working fluid;

precooling coil means in said return air flow for exchanging thermal energy between the return air flow and the working fluid;

reheat coil means in said supply air flow for exchanging thermal energy between the working fluid and the supply air flow;

heat exchange means for exchanging thermal energy between the compressible fluid and the working fluid;

fluid pump means for motivating a flow of the working fluid through said precooling coil means, said reheat coil means, and said heat exchange means; and,

regulating means for regulating said working fluid flow through said precooling and reheat coil.

2. A moisture control apparatus according to claim 1 further comprising fluid conduit means containing the working fluid therein for containedly directing the working fluid through a series arrangement of said precooling coil means, said heat exchange means, and said reheat coil means.

3. A moisture control apparatus according to claim 2 wherein said regulating means comprises a control valve connected to said fluid conduit means in said series arrangement.

4. A moisture control apparatus according to claim 3 further comprising bypass conduit means, connected to said control valve in parallel with said heat exchange means and in parallel with a series combination of said precooling coil means and said reheat coil means, for selectively circulating a first portion of the working fluid as a bypass flow through said series combination of said precooling coil means and said reheat coil means.

5. A moisture control apparatus according to claim 4 wherein said control valve comprises a first input port connected to said fluid conduit means for receiving a first flow of said working fluid from said heat exchange means, a second input port connected to said bypass conduit means for receiving said bypass flow of said working fluid from said precooling coil means, an output port connected to said fluid conduit means for selectively exhausting said first and bypass flows from said control valve to said reheat coil means, and valving means for selectively metering said first and bypass flows through said control valve as a metered flow.

6. A moisture control apparatus according to claim 5 wherein said fluid pump means comprises a variable speed drive fluid pump.

7. A moisture control apparatus according to claim 6 further comprising control means operatively associated with said control valve and said variable speed drive fluid pump for sensing moisture in said supply air flow and for maintaining said sensed moisture at a predetermined set point by regulating i) said valving means to selectively meter said first and bypass flows, and ii) said variable speed drive fluid pump to motivate the metered flow through said series combination of said reheat coil means and said precooling coil means.

8. A moisture control apparatus according to claim 7 further comprising thermal energy storage unit means, connected to said fluid conduit means and operatively associated with said working fluid and said heat exchange means, for recovering and storing thermal energy from said compressible fluid and selectively delivering the stored thermal energy to said working fluid.

9. A humidity control apparatus for use with a cooling coil means for selectively absorbing thermal energy from a return airstream as a cooled airstream, the apparatus comprising:

precooling coil means disposed upstream of said cooling coil means for selectively precooling said return airstream;

reheat coil means disposed downstream of said cooling coil means for selectively reheating the cooled airstream flowing from said cooling coil means;

sensing means disposed downstream of said reheat coil means for sensing the relative humidity of the reheated airstream flowing from said reheat coil

means and generating a humidity signal reflective of said sensed relative humidity;

conduit means connecting said precooling and reheat coil means for communicating a working fluid flow through a closed loop circuit comprising said precooling coil means and said reheat coil means;

pump means for controlledely motivating the working fluid flow through said closed loop circuit at a controlled flow rate responsive to said humidity signal; and,

means for selectively introducing thermal energy into said working fluid flow responsive to i) said humidity signal and ii) said pump means motivating the working fluid flow at a predefined maximum controlled flow rate.

10. The humidity control apparatus according to claim 9 wherein said means for selectively introducing thermal energy into said working fluid flow comprises:

thermal energy storage means for storing thermal energy;

means disposed downstream of said pump means and in said closed loop circuit for dividing said working fluid flow into at least two partial parallel fluid flows comprising i) a bypass fluid flow and ii) a heated fluid flow passing through said thermal energy storage means; and,

metering means disposed in said closed loop circuit upstream of said reheat coil means for receiving said bypass fluid flow at a first inlet port and said heated fluid flow at a second inlet port and selectively metering the received fluid flows for exhaust at an exhaust port connected to said conduit means.

11. A method of heating a supply airstream into a conditioned space from a return airstream and downstream of a cooling coil of a typical air conditioning system operating in a space dehumidification mode for dehumidifying the conditioned space responsive to a humidity sensor in the conditioned space, the method comprising the steps of:

storing heat energy in a thermal energy storage tank;

sensing the temperature of the conditioned space;

selectively circulating a working fluid at a controlled flow rate from a precooling coil in the return airstream of the air conditioning system directly to a reheat coil in said supply airstream when said sensed temperature is below a first predetermined set point; and,

selectively circulating the working fluid from the precooling coil through the thermal energy storage tank and the reheat coil in series combination when said controlled flow rate is at a predetermined maximum rate and said sensed temperature is below the first predetermined set point.

12. The method according to claim 11 further comprising the steps of recovering waste thermal energy from said air conditioning system and storing the recovered waste thermal energy in said thermal energy storage tank.

13. A method of controlling the relative humidity of a supply airstream into a conditioned space from a return airstream and downstream of a cooling coil of a typical air conditioning system operating in a space

cooling mode for cooling the conditioned space responsive to a dry bulb temperature sensor in the conditioned space, the method comprising the steps of:

storing heat energy in a thermal energy storage tank;

sensing the relative humidity of the supply airstream;

selectively circulating a working fluid at a controlled flow rate from a precooling coil in the return airstream of the air conditioning system directly to a reheat coil in said supply airstream when said sensed relative humidity is above a first predetermined set point; and,

selectively circulating the working fluid from the precooling coil through the thermal energy storage tank and the reheat coil in series combination when said controlled flow rate is at a predetermined maximum rate and said sensed relative humidity is above the first predetermined set point.

14. The method according to claim 13 further comprising the steps of recovering waste thermal energy from said air conditioning system and storing the recovered waste thermal energy in said thermal energy storage tank.

15. An apparatus for controlling the relative humidity of an airstream supplied into a temperature conditioned space downstream of an air conditioning cooling coil, the apparatus comprising:

control means for controlling the apparatus according to a predetermined control method;

precooling means in said airstream for precooling the airstream upstream of said cooling coil;

reheating means in said airstream for reheating the airstream downstream of said cooling coil;

connecting means for connecting said precooling means and said reheating means in a series closed loop;

humidity sensing means in the airstream downstream of the reheating means and connected to said control means for i) sensing the relative humidity of the airstream between said reheating means and said temperature conditioned space and ii) generating a humidity signal representing the sensed humidity;

working fluid pump means responsive to said control means for selectively pumping a working fluid through said series closed loop when said humidity signal is at a predetermined level, the working fluid pump means having an inherent maximum pumping rate;

thermal energy storage tank means in fluid communication with said connecting means for storing thermal energy therein; and,

regulating means connected to said connecting means and said thermal energy storage tank means responsive to said control means for selectively regulating a flow of said working fluid through said series closed loop and said thermal energy storage tank means when said working fluid pump means is pumping at said maximum pumping rate and said humidity signal is at said predetermined level.

16. An apparatus for controlling the temperature of an airstream supplied into a moisture conditioned space downstream of an air conditioning cooling coil, the apparatus comprising:

control means for controlling the apparatus according to a predetermined control method;

precooling means in said airstream for precooling the airstream upstream of said cooling coil;

reheating means in said airstream for reheating the airstream downstream of said cooling coil;

connecting means for connecting said precooling means and said reheating means in a series closed loop;

temperature sensing means in the conditioned space downstream of the reheating means and connected to said control means for i) sensing the temperature of the conditioned space downstream of said reheating means and ii) generating a temperature signal representing the sensed temperature;

working fluid pump means responsive to said control means for selectively pumping a working fluid through said series closed loop when said temperature signal is at a predetermined level, the working fluid pump means having an inherent maximum pumping rate;

thermal energy storage tank means in fluid communication with said connecting means for storing thermal energy therein; and,

regulating means connected to said connecting means and said thermal energy storage tank means responsive to said control means for selectively regulating a flow of said working fluid through said series closed loop and said thermal energy storage tank means when said working fluid pump means is pumping at said maximum pumping rate and said temperature signal is at said predetermined level.

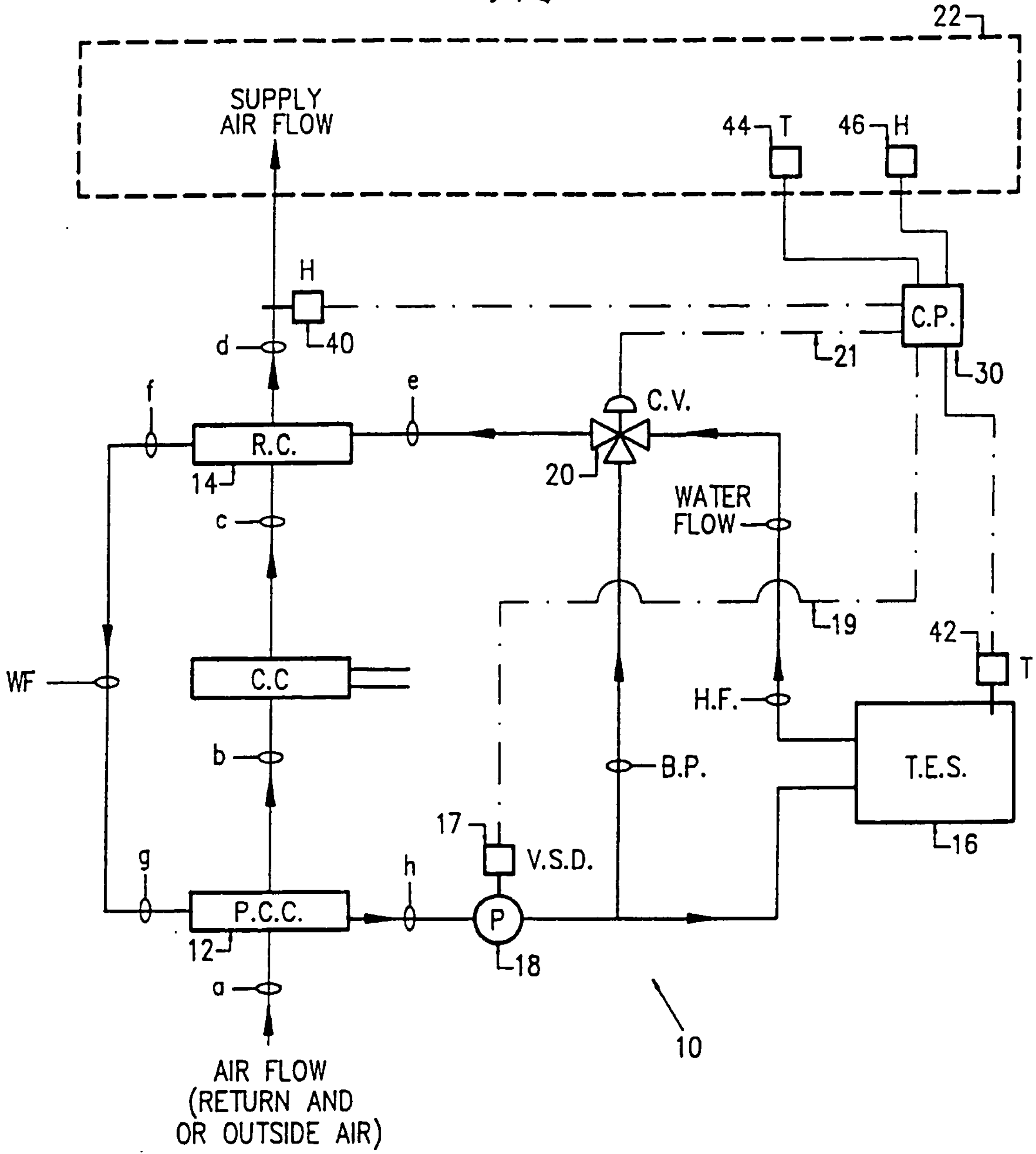

22
SUPPLY
AIR FLOW
44 T
46 H
H
40
C.P.
30
d
21
f
e
C.V.
R.C.
20
14
WATER
FLOW
c
19
42
T
WF
C.C
H.F.
T.E.S.
b
B.P.
17
V.S.D.
16
g
h
P.C.C.
P
12
18
a
10
AIR FLOW
(RETURN AND
OR OUTSIDE AIR)

FIG.-1

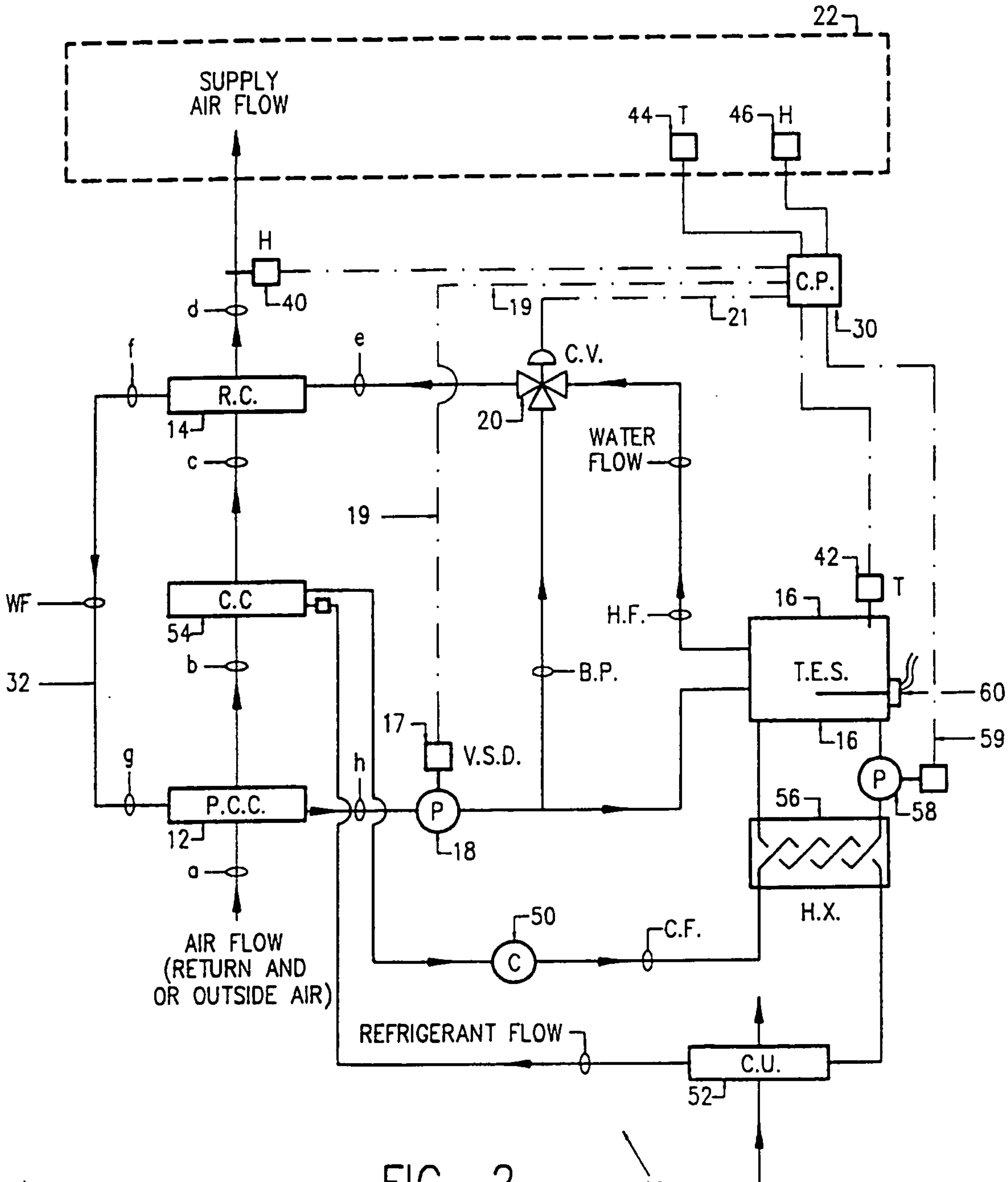

22
SUPPLY
AIR FLOW
44
T
46
H
H
40
C.P.
30
19
21
d
f
e
C.V.
R.C.
14
20
WATER
FLOW
c
19
42
T
WF
C.C
54
16
H.F.
32
b
B.P.
T.E.S.
60
17
V.S.D.
16
59
g
h
P
P
56
58
P.C.C.
12
18
H.X.
a
50
C
C.F.
AIR FLOW
(RETURN AND
OR OUTSIDE AIR)
REFRIGERANT FLOW
C.U.
52
10

FIG.-2

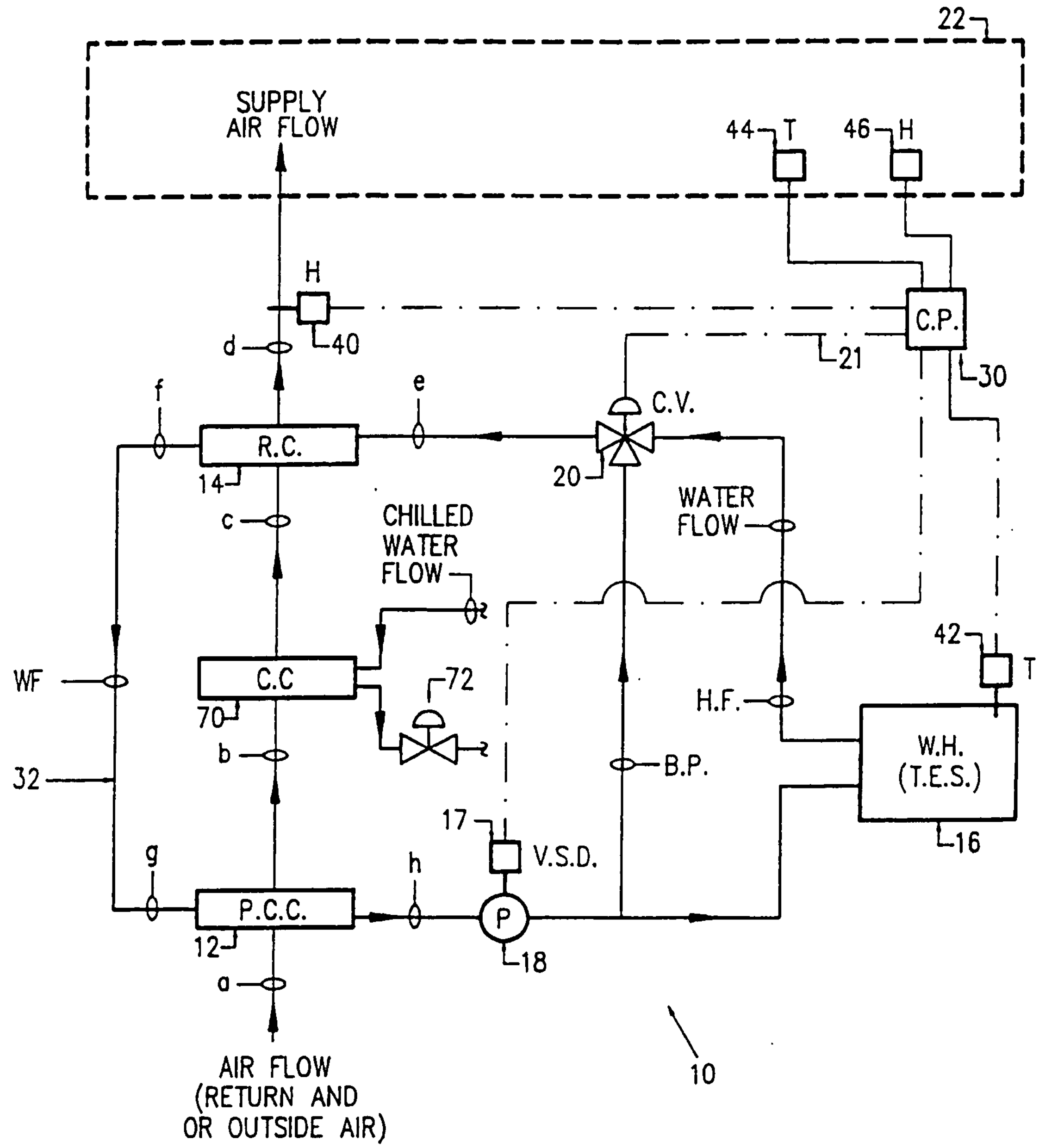

22
SUPPLY AIR FLOW
44 T
46 H
H
40
C.P.
30
21
d
f
e
C.V.
R.C.
14
20
c
CHILLED WATER FLOW
WATER FLOW
WF
C.C
70
72
42
T
H.F.
W.H. (T.E.S.)
16
32
b
B.P.
17
V.S.D.
g
h
P.C.C.
12
P
18
a
10
AIR FLOW (RETURN AND OR OUTSIDE AIR)

FIG.-3

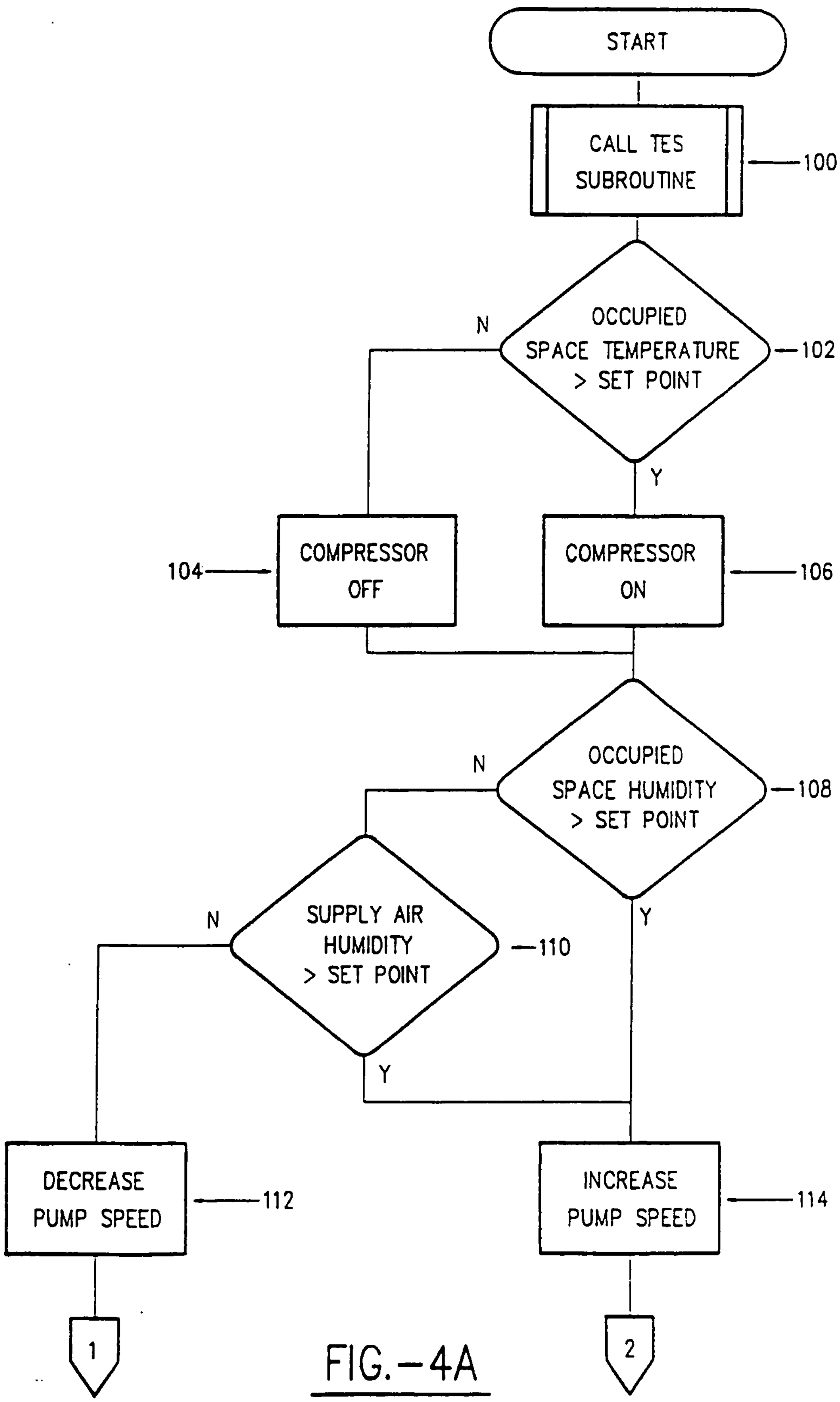

START
CALL TES SUBROUTINE
100
OCCUPIED SPACE TEMPERATURE > SET POINT
102
N
Y
COMPRESSOR OFF
104
COMPRESSOR ON
106
OCCUPIED SPACE HUMIDITY > SET POINT
108
N
Y
SUPPLY AIR HUMIDITY > SET POINT
110
N
Y
DECREASE PUMP SPEED
112
INCREASE PUMP SPEED
114
1
2

FIG.-4A

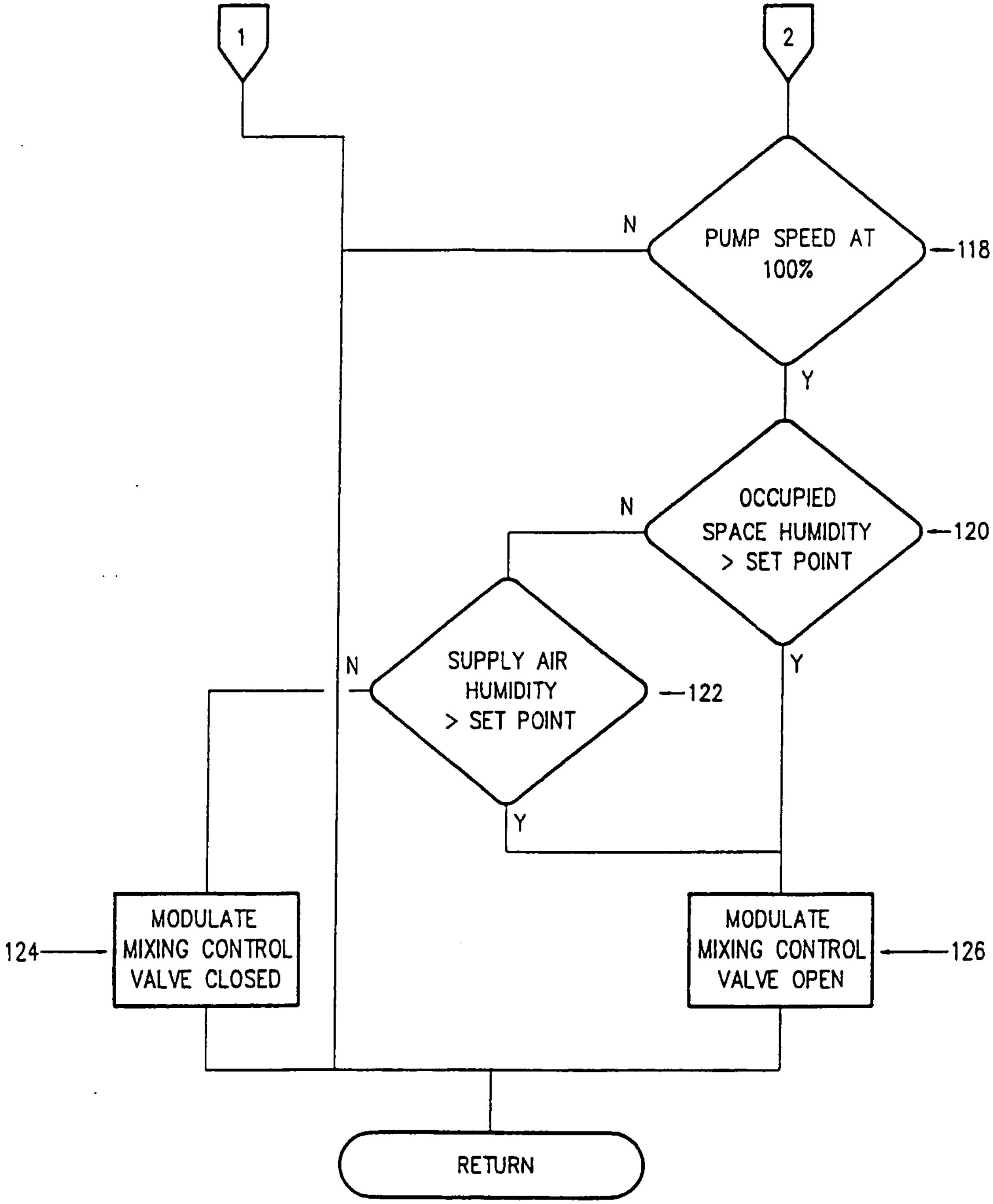

1
2
PUMP SPEED AT 100%
N
Y
118
OCCUPIED SPACE HUMIDITY > SET POINT
N
Y
120
SUPPLY AIR HUMIDITY > SET POINT
N
Y
122
124
MODULATE MIXING CONTROL VALVE CLOSED
MODULATE MIXING CONTROL VALVE OPEN
126
RETURN

FIG.–4B

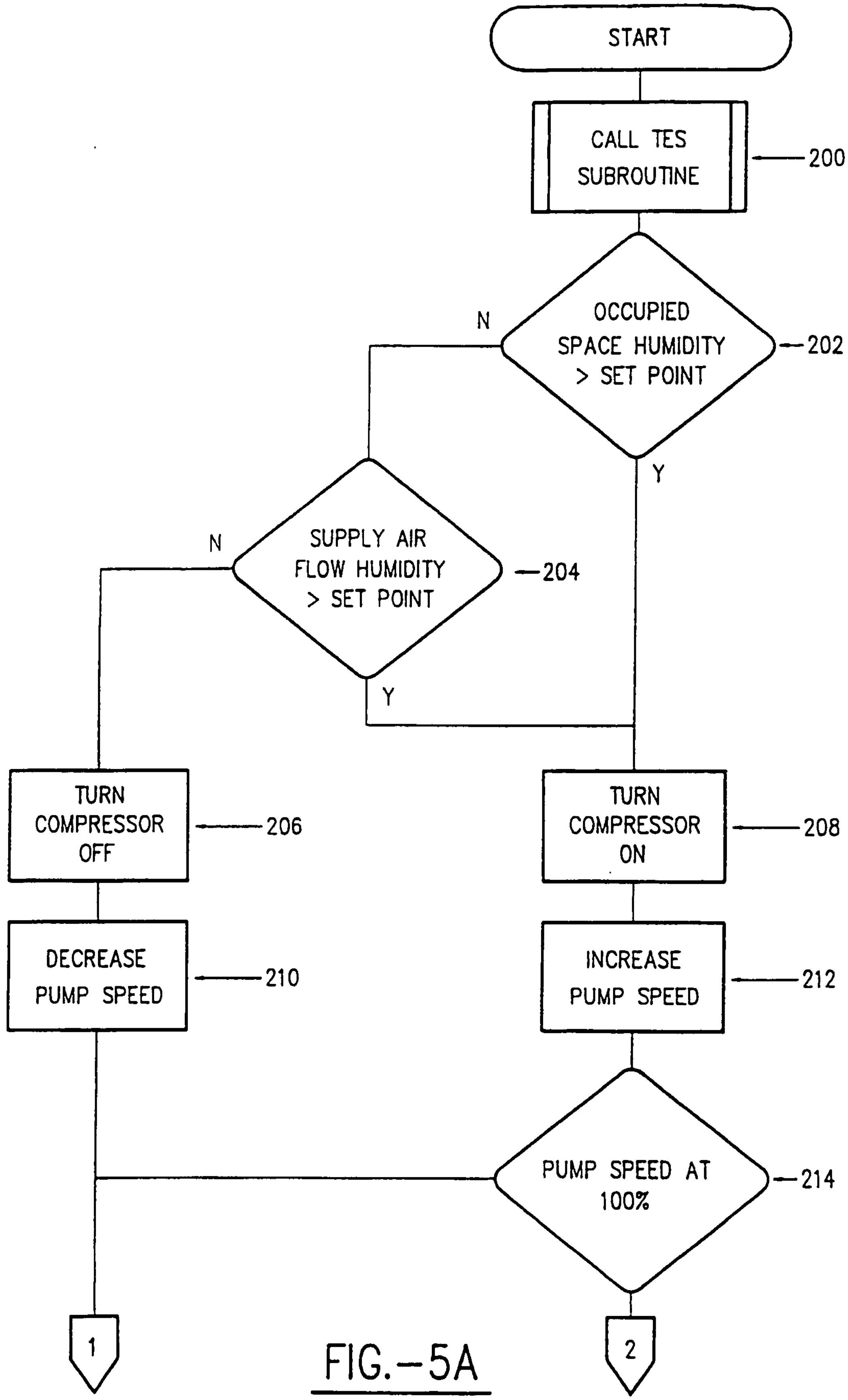

START
CALL TES SUBROUTINE
200
OCCUPIED SPACE HUMIDITY > SET POINT
202
N
Y
SUPPLY AIR FLOW HUMIDITY > SET POINT
204
N
Y
TURN COMPRESSOR OFF
206
TURN COMPRESSOR ON
208
DECREASE PUMP SPEED
210
INCREASE PUMP SPEED
212
PUMP SPEED AT 100%
214
1
2

FIG.—5A

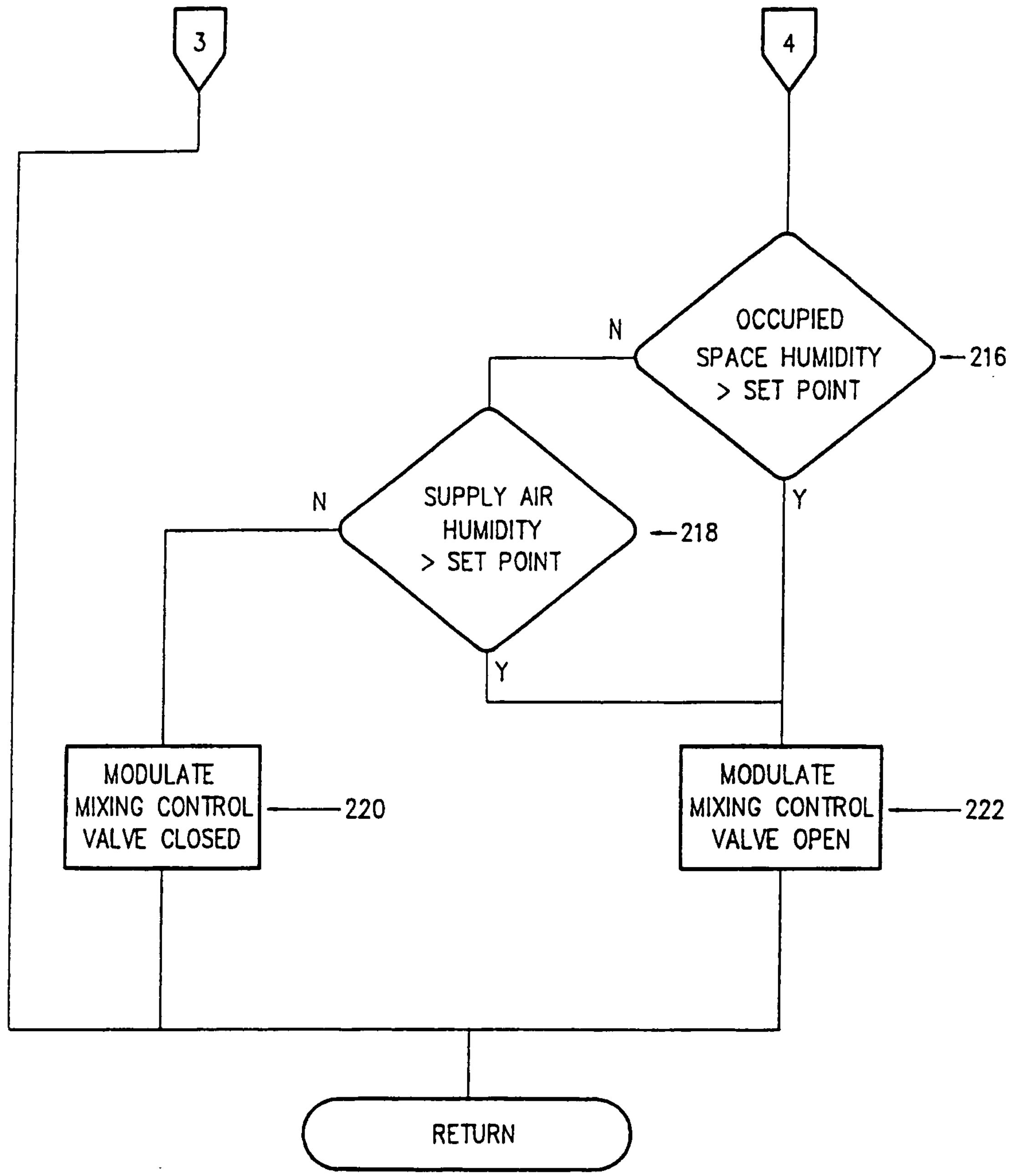

3
4
OCCUPIED SPACE HUMIDITY > SET POINT
216
N
Y
SUPPLY AIR HUMIDITY > SET POINT
218
N
Y
MODULATE MIXING CONTROL VALVE CLOSED
220
MODULATE MIXING CONTROL VALVE OPEN
222
RETURN

FIG.-5B

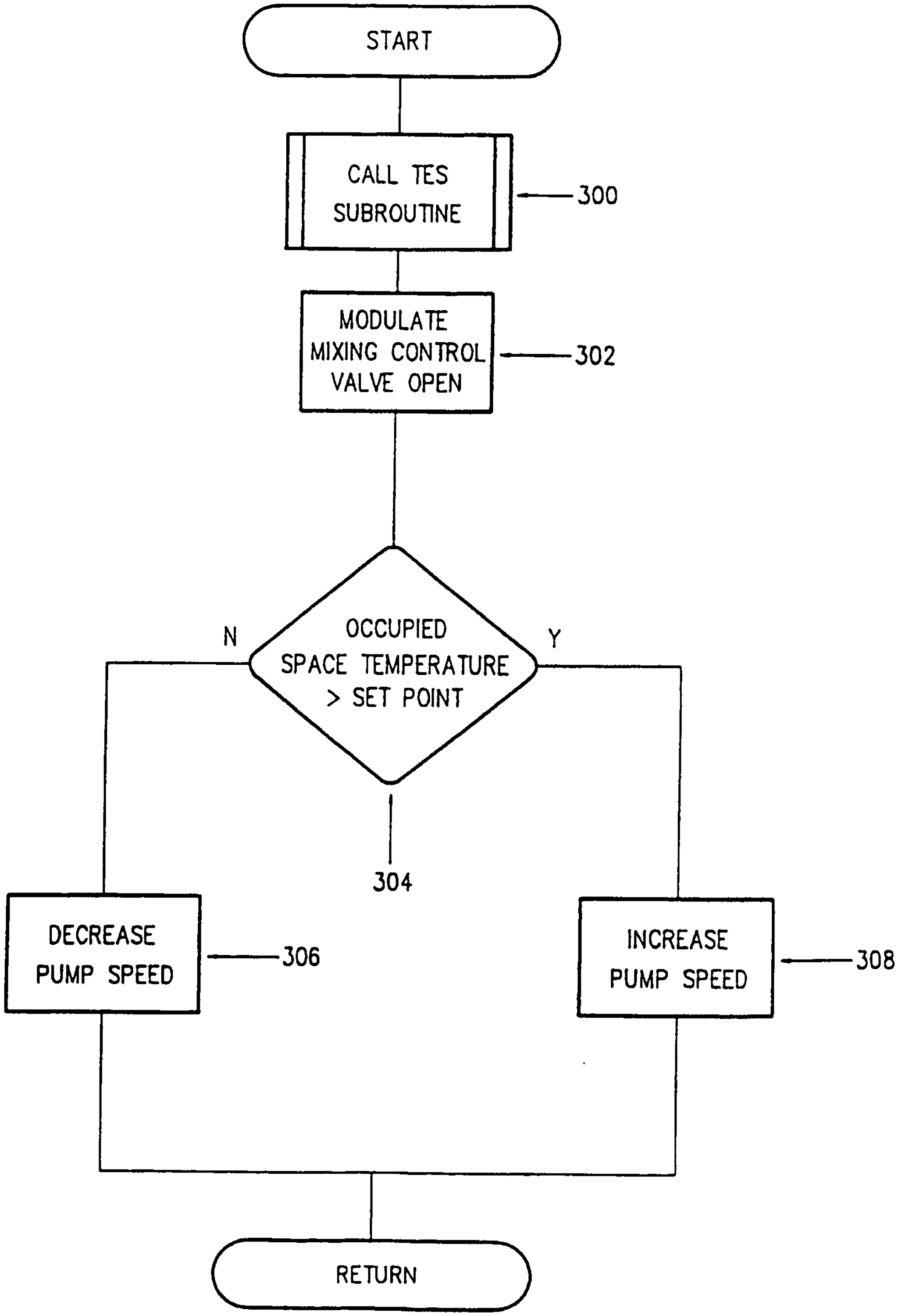

START
CALL TES SUBROUTINE
300
MODULATE MIXING CONTROL VALVE OPEN
302
OCCUPIED SPACE TEMPERATURE > SET POINT
N
Y
304
DECREASE PUMP SPEED
306
INCREASE PUMP SPEED
308
RETURN

FIG -6

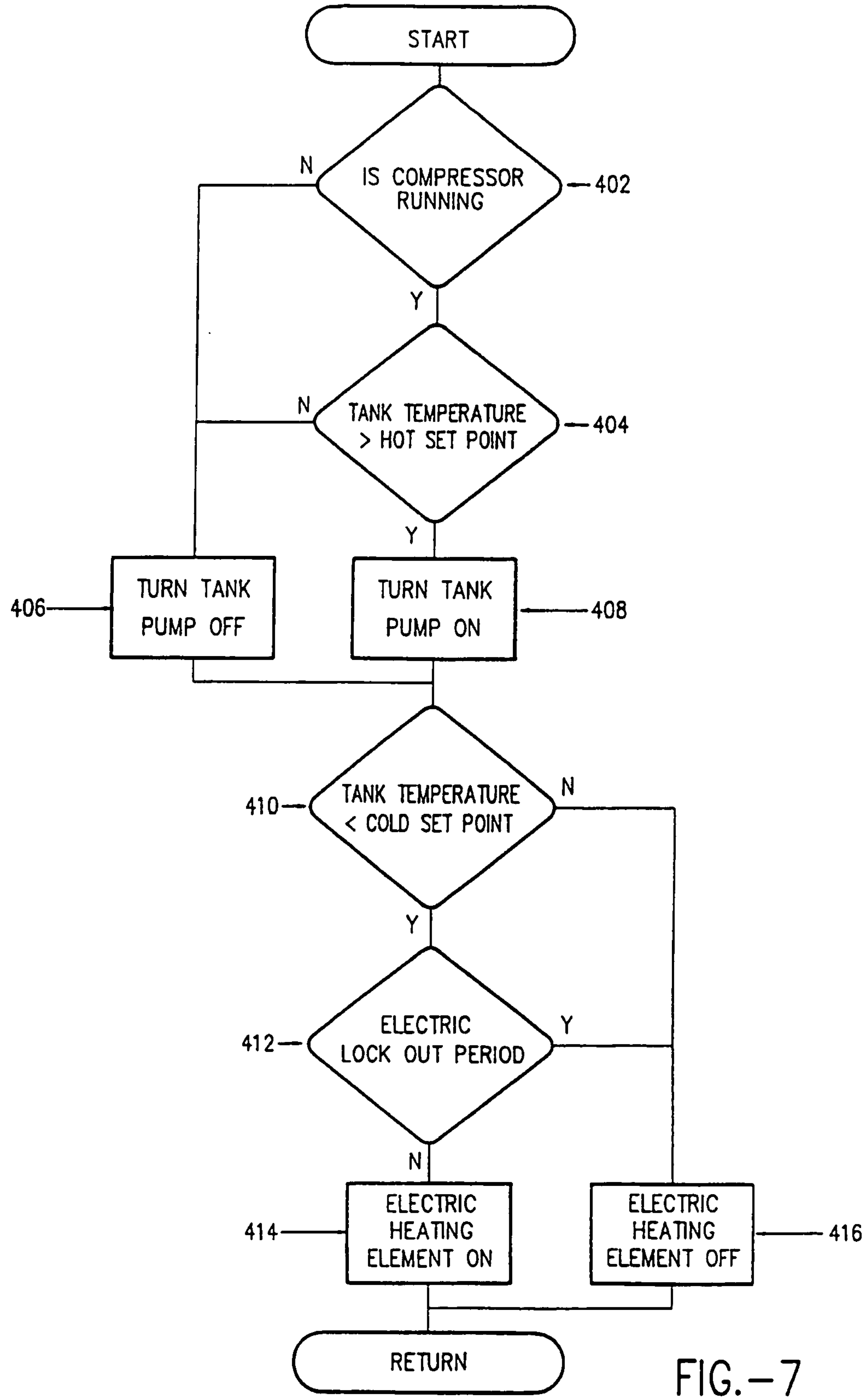

START
IS COMPRESSOR RUNNING
402
N
Y
TANK TEMPERATURE > HOT SET POINT
404
N
Y
TURN TANK PUMP OFF
406
TURN TANK PUMP ON
408
TANK TEMPERATURE < COLD SET POINT
410
N
Y
ELECTRIC LOCK OUT PERIOD
412
Y
N
ELECTRIC HEATING ELEMENT ON
414
ELECTRIC HEATING ELEMENT OFF
416
RETURN

FIG.-7

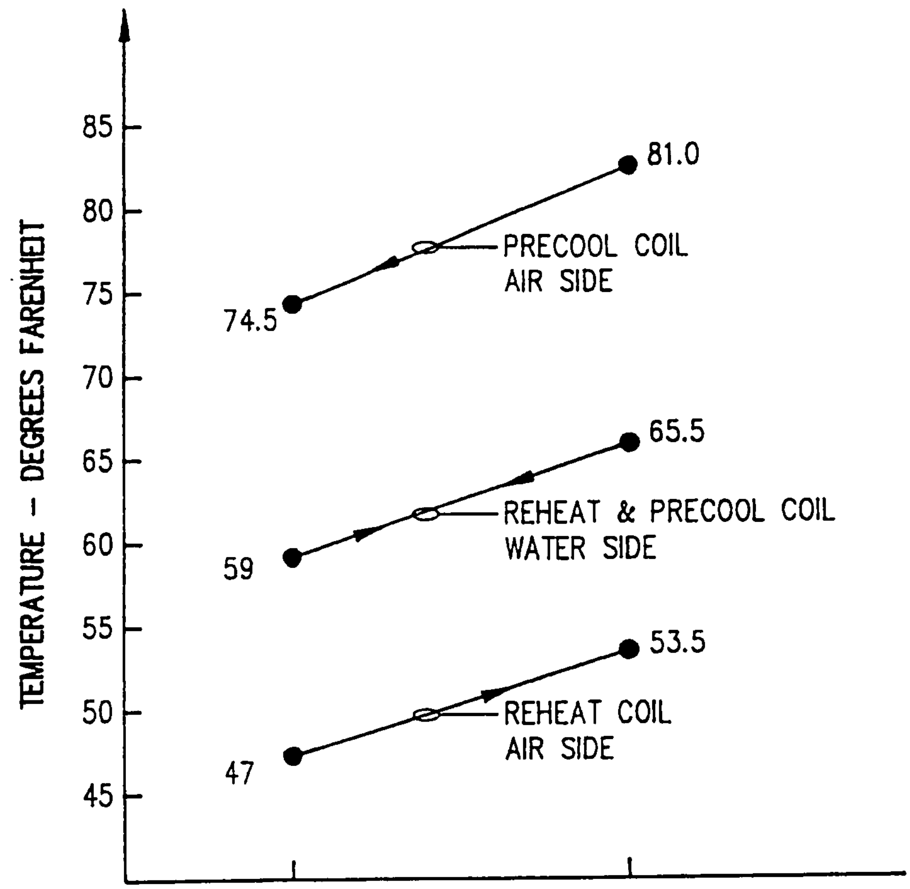

85
80
75
70
65
60
55
50
45
TEMPERATURE - DEGREES FARENHEIT
81.0
PRECOOL COIL
AIR SIDE
74.5
65.5
REHEAT & PRECOOL COIL
WATER SIDE
59
53.5
REHEAT COIL
AIR SIDE
47

FIG.-8

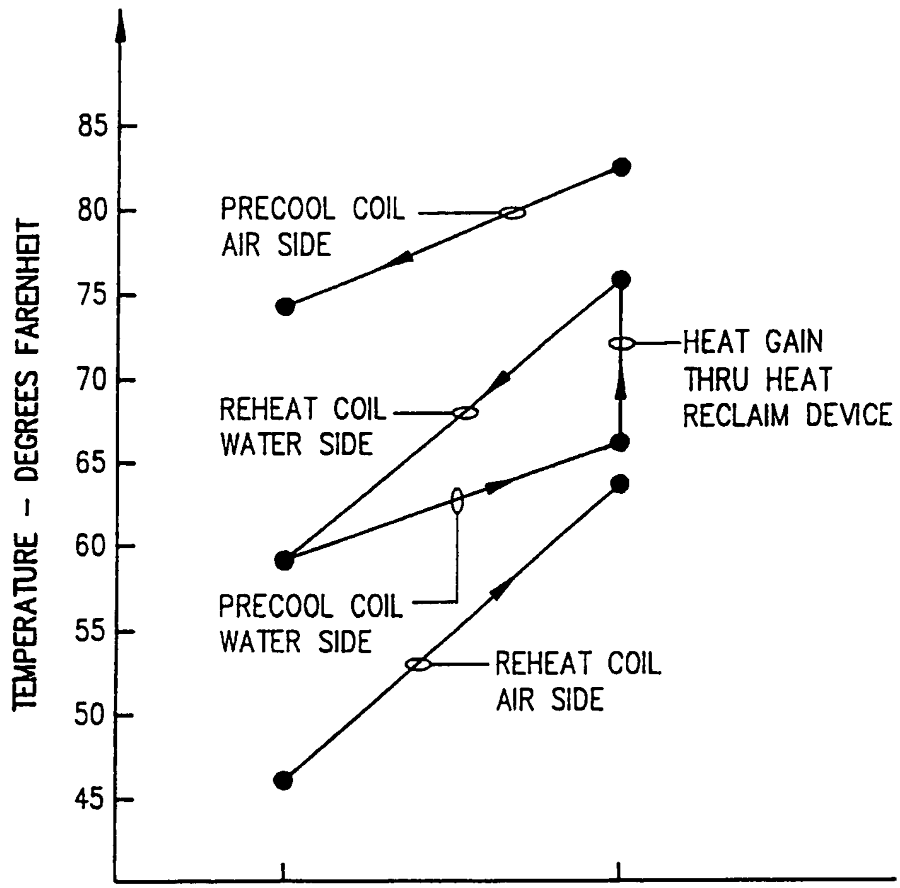

TEMPERATURE - DEGREES FARENHEIT
85
80
75
70
65
60
55
50
45
PRECOOL COIL
AIR SIDE
REHEAT COIL
WATER SIDE
PRECOOL COIL
WATER SIDE
REHEAT COIL
AIR SIDE
HEAT GAIN
THRU HEAT
RECLAIM DEVICE

FIG.-9

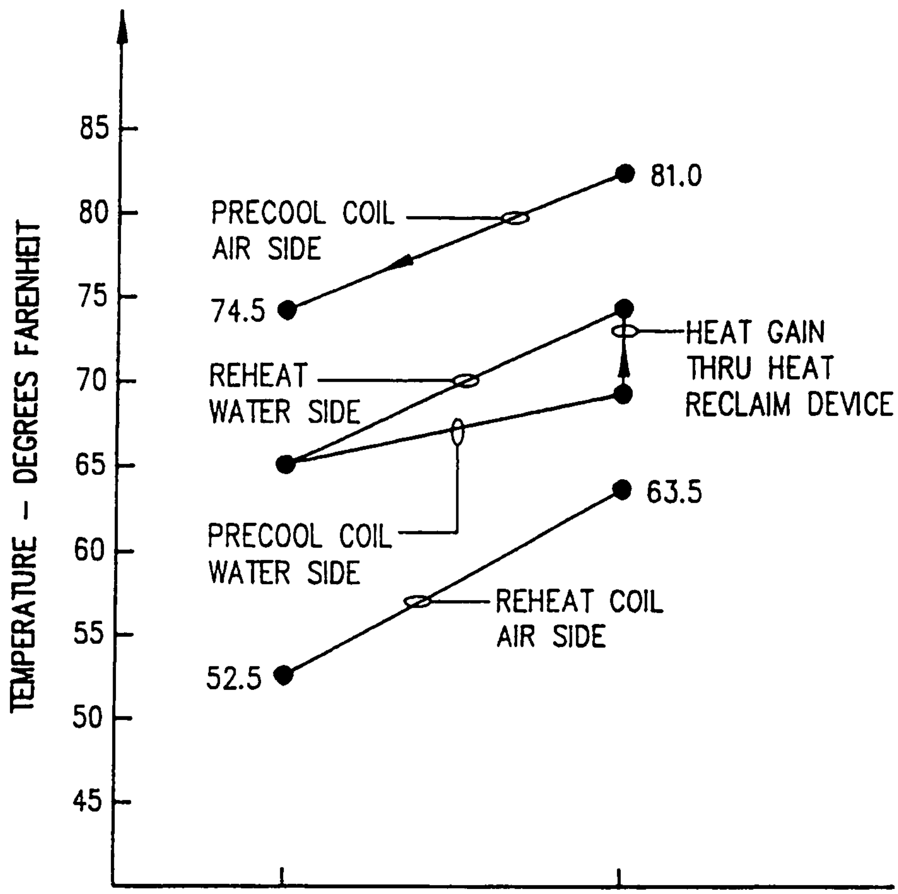

85
80
75
70
65
60
55
50
45
TEMPERATURE - DEGREES FARENHEIT
81.0
PRECOOL COIL
AIR SIDE
74.5
HEAT GAIN
THRU HEAT
RECLAIM DEVICE
REHEAT
WATER SIDE
63.5
PRECOOL COIL
WATER SIDE
REHEAT COIL
AIR SIDE
52.5

FIG.-10

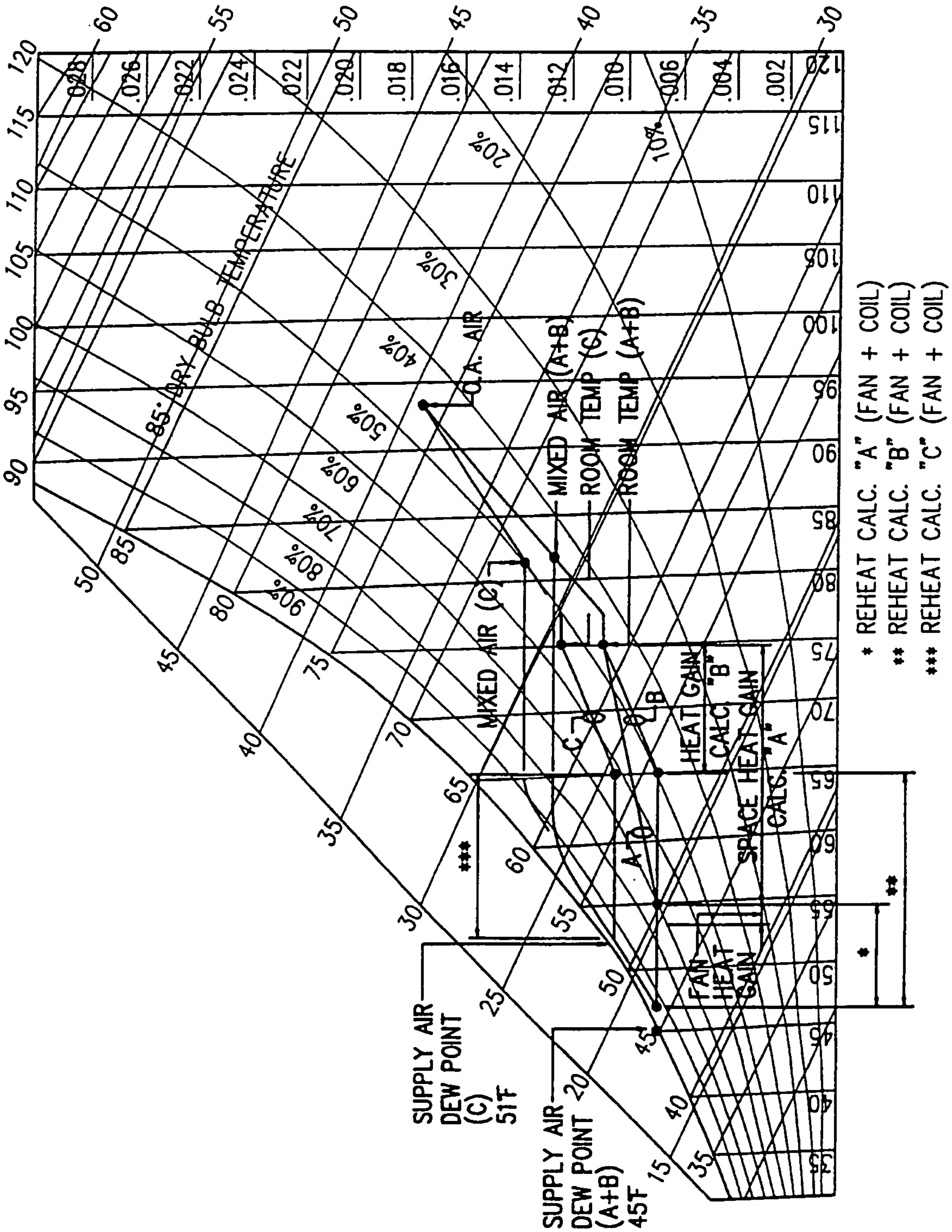
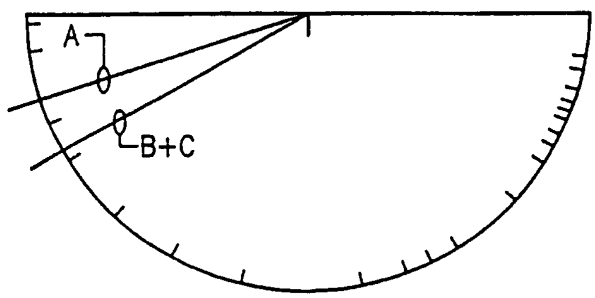

85° DRY BULB TEMPERATURE
O.A. AIR
MIXED AIR (A+B)
ROOM TEMP (C)
ROOM TEMP (A+B)
MIXED AIR (C)
HEAT GAIN CALC. "B"
SPACE HEAT GAIN CALC. "A"
FAN HEAT GAIN
SUPPLY AIR DEW POINT (C) 51F
SUPPLY AIR DEW POINT (A+B) 45F
* REHEAT CALC. "A" (FAN + COIL)
** REHEAT CALC. "B" (FAN + COIL)
*** REHEAT CALC. "C" (FAN + COIL)

FIG.-11A

SEA LEVEL PSYCHROMETRIC CHART



PROTRACTOR
SENSIBLE HEAT
TOTAL HEAT

FIG.–11B

INTERNATIONAL SEARCH REPORT

PCT/US92/09818

A. CLASSIFICATION OF SUBJECT MATTER

IPC(5) :F25D 17/06; F25B 29/00
US CL :62/90, 62/173
According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

U.S. : 62/90, 62/173 62/176.5

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	US,A, 2,286,605 (Crawford) 16 June 1942 See Fig. 1.	1-16
A	US,A, 2,291,029 (Everetts, Jr.) 28 July 1942 See Fig. 2, 50,130,114.	1-16
A	US,A, 4,271,678 (Liebert) 09 June 1981 See Fig. 1.	1-16

☐ Further documents are listed in the continuation of Box C. ☐ See patent family annex.

* Special categories of cited documents:

"A" document defining the general state of the art which is not considered to be part of particular relevance

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Date of the actual completion of the international search	Date of mailing of the international search report
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